March 15, 1966 R. G. HOWARD ET AL 3,240,124

HYDRAULIC SERVOMECHANISM

Filed June 12, 1963 4 Sheets-Sheet 1

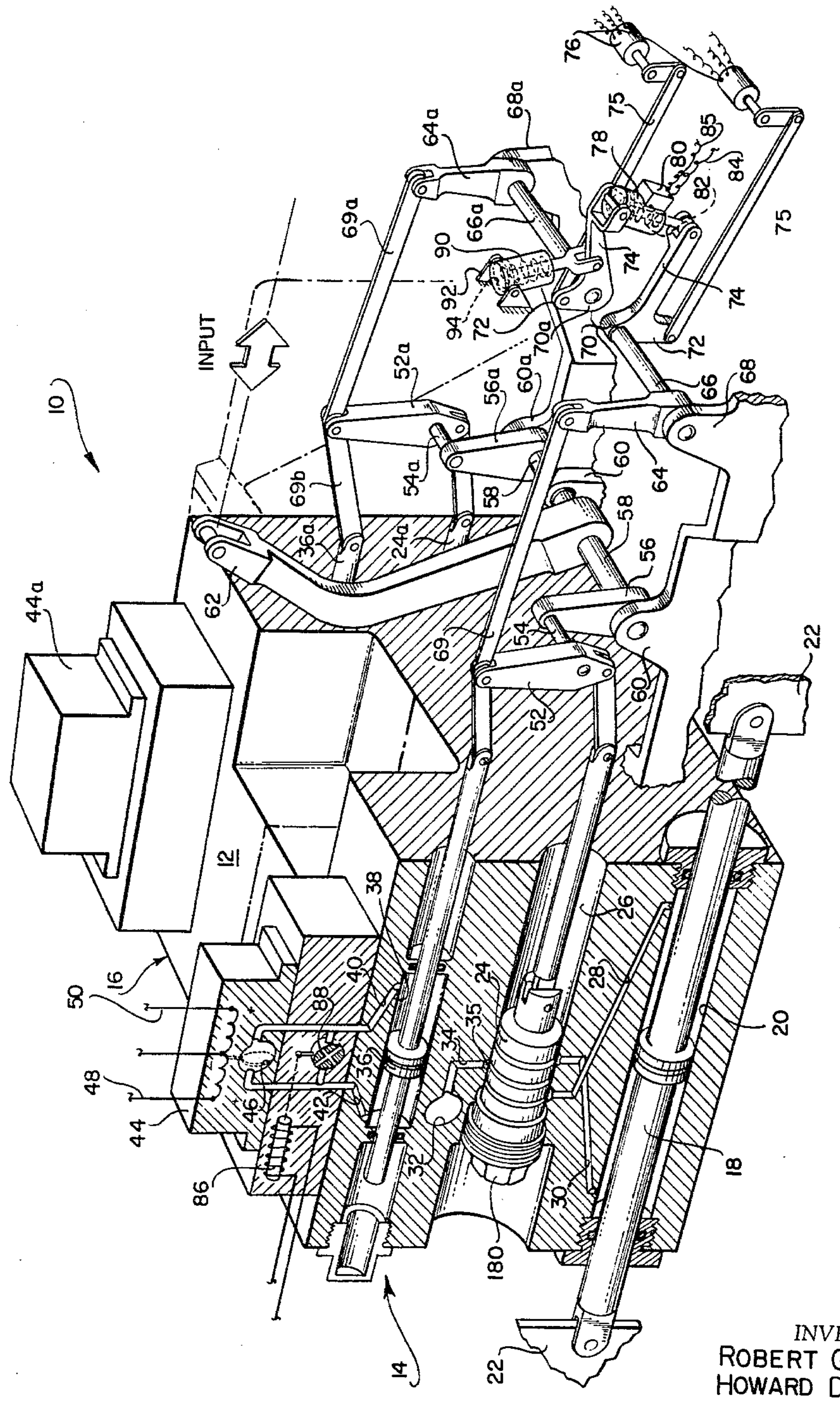

FIG. 1

INVENTORS
ROBERT G. HOWARD
HOWARD D. KURTZ
By George C. Sullivan
Agent

March 15, 1966 R. G. HOWARD ET AL 3,240,124

HYDRAULIC SERVOMECHANISM

Filed June 12, 1963 4 Sheets-Sheet 2

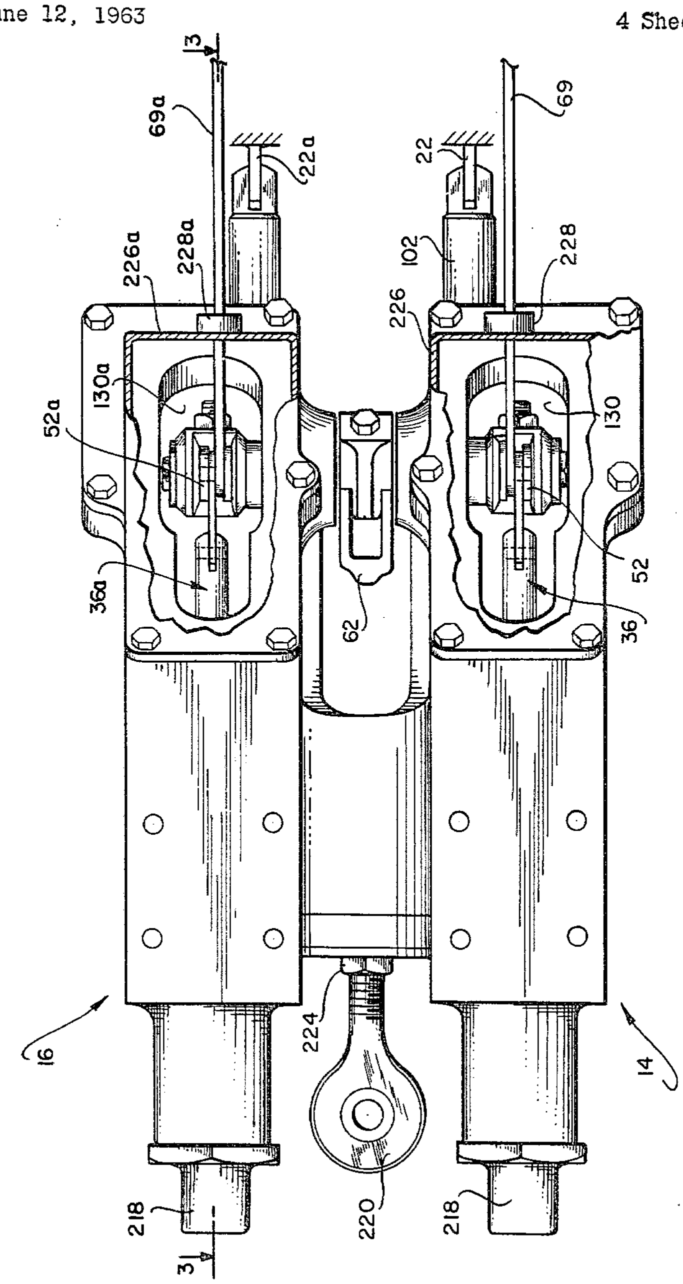

FIG. 2

INVENTORS
ROBERT G. HOWARD
HOWARD D. KURTZ
By George C. Sullivan
Agent

March 15, 1966 R. G. HOWARD ET AL 3,240,124

HYDRAULIC SERVOMECHANISM

Filed June 12, 1963 4 Sheets-Sheet 3

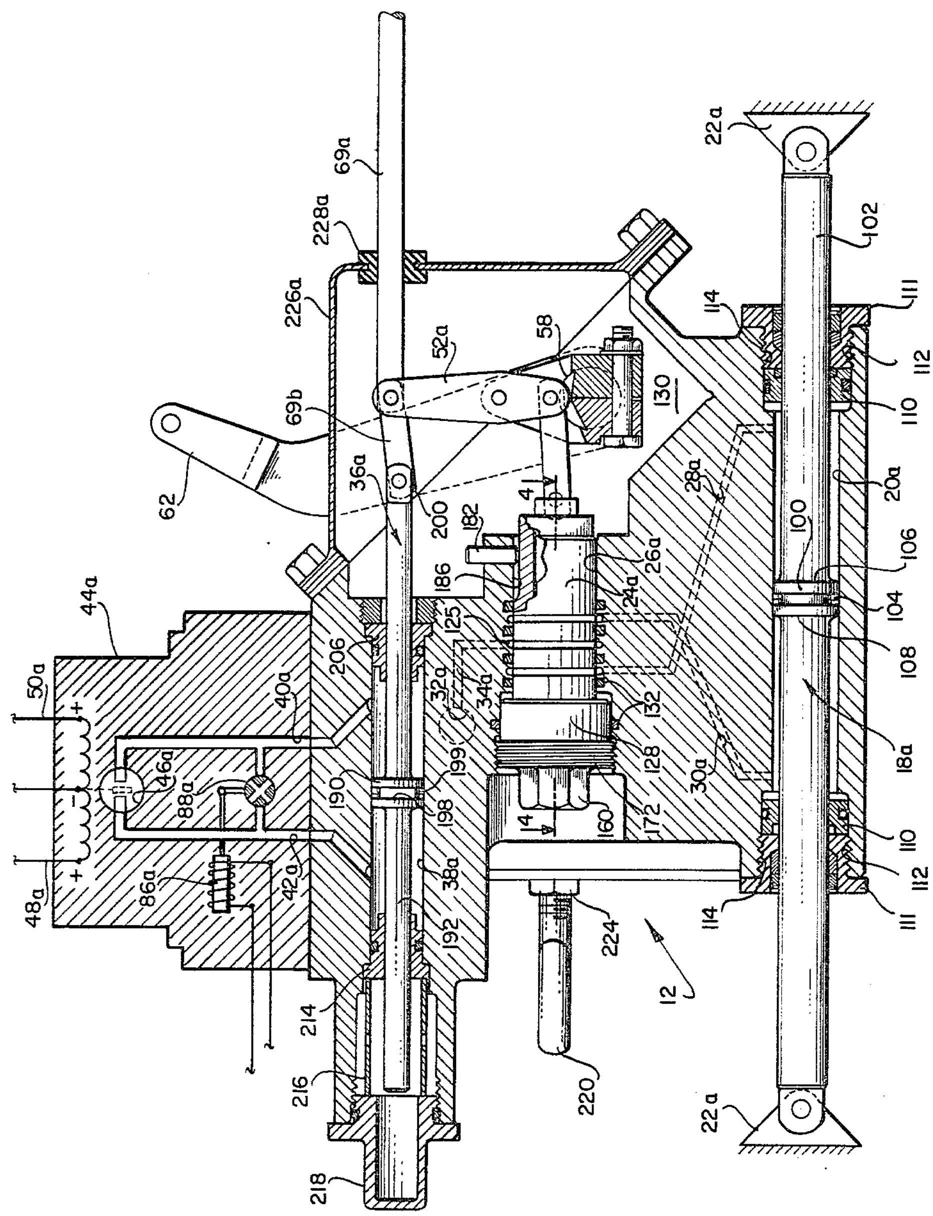

FIG_3

INVENTORS
ROBERT G. HOWARD
HOWARD D. KURTZ

By George Sullivan
Agent

March 15, 1966 R. G. HOWARD ET AL 3,240,124

HYDRAULIC SERVOMECHANISM

Filed June 12, 1963 4 Sheets-Sheet 4

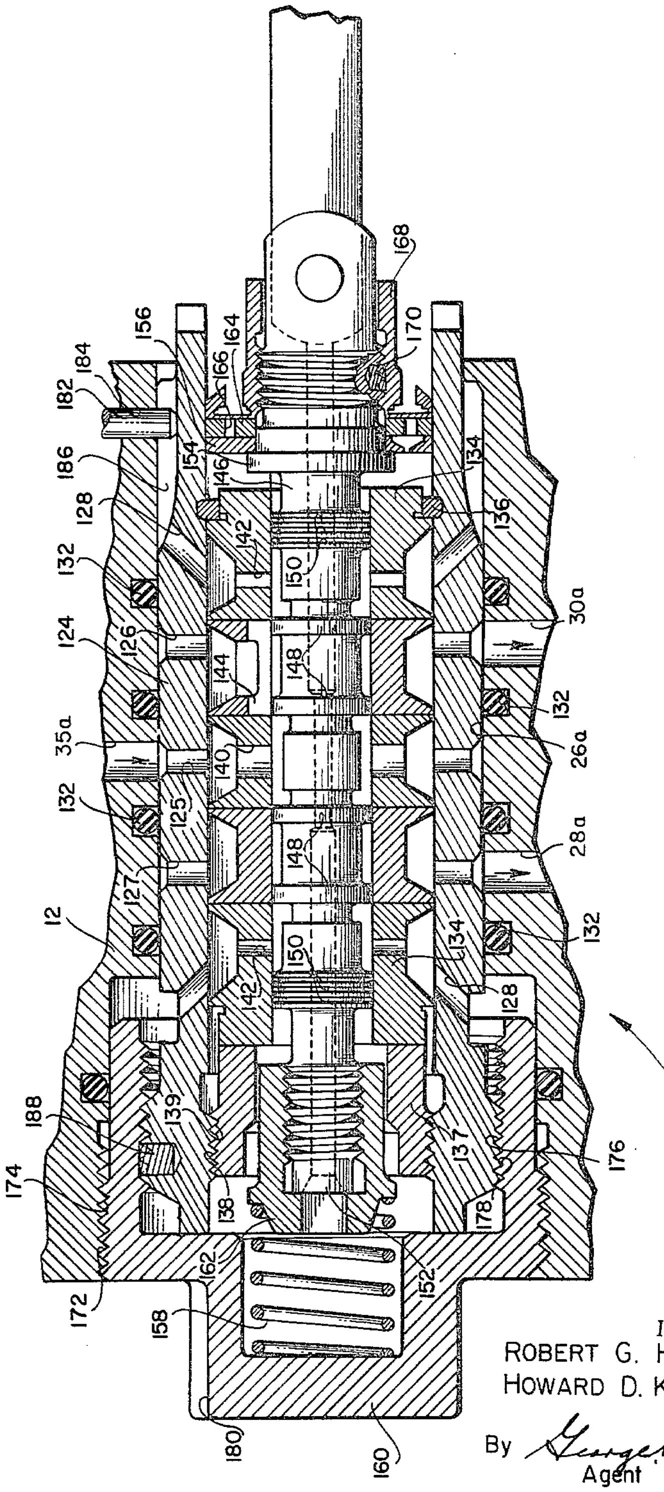

FIG. 4

INVENTORS
ROBERT G. HOWARD
HOWARD D. KURTZ

By George Sullivan
Agent

United States Patent Office 3,240,124
Patented Mar. 15, 1966

3,240,124
HYDRAULIC SERVOMECHANISM

Robert G. Howard, Northridge, and Howard D. Kurtz, Los Angeles, Calif., assignors to Lockheed Aircraft Corporation, Burbank, Calif.

Filed June 12, 1963, Ser. No. 287,283
9 Claims. (Cl. 91—1)

This invention relates to a hydraulic servomechanism and more particularly to a dual-parallel, hydraulic servomechanism for aircraft which encompasses a completely independent hydraulic system in each half of a single unit in such a manner that failure or malfunction of one of the systems leaves the other system free to operate along with no loss of aircraft control.

Servomechanisms may be used to position aerodynamic control surfaces on conventional aircraft and to position the thrust levers to vary the thrust output of engines on VTOL aircraft. It is known to increase the reliability of such servomechanisms by providing dual-tandem servomechanisms in which modulating pistons and power pistons from two systems are each mounted coaxially, as with two cylinders having a common piston rod. The tandem or coaxial arrangement insures a fixed relationship between the duplicated systems. Although generally satisfactory, such systems do have certain drawbacks.

One drawback resides in the fact that it is difficult to detect a malfunction in one half of a dual-tandem system.

Another drawback resides in the fact that it is difficult to provide a satisfactory seal between each half of the system to prevent transfer of hydraulic fluid from one half of the system to the other half, especially when the latter half of the pair has a reduced, hydraulic supply pressure.

Yet another drawback resides in the fact that it is difficult to detect a malfunction in the electrical systems which automatically control the servomechanism.

In view of the foregoing factors and conditions characteristic of servomechanisms having two-system reliability, it is a primary object of the present invention to provide a new and improved servomechanism not subject to the disadvantages enumerated above and having a dual-parallel arrangement in which two independent, servo units are built into a single unit.

Another object of the present invention is to provide a redundant servomechanism having two independent modulating pistons which are linked to a coaxial shaft in such a manner that detection of non-uniformity of action between the two pistons is readily ascertainable.

Yet another object of the present invention is to provide means for detecting a malfunction in a dual-parallel servomechanism.

A further object of the present invention is to provide a dual-parallel servomechanism having new and useful means for matching the performance of separate spool valves.

A still further object of the present invention is to provide a vernier-type adjustment which permits accurate, rigid positioning of inlet ports of a hydraulic servomechanism.

Yet another object of the present invention is to provide a redundant servomechanism for aircraft which employs a completely independent hydraulic system in each half of a consolidated unit in such a manner that failure or malfunction of one of the systems permits the other system to operate individually with no loss of aircraft control.

Another object of the present invention is to provide a dual-parallel servomechanism which will remain in automatic operation even though the electrical system controlling the operation of one half of the servomechanism is interrupted.

Another object of the present invention is to provide a dual-parallel servomechanism for aircrat having means which will signal the pilot of the aircraft to return to manual controls when the electrical system serving either half of the dual servomechanism malfunctions.

According to the present invention, a dual-parallel hydraulic servomechanism is provided in which a single servomechanism unit encompasses two completely independent hydraulic and electrical systems. In case of failure or malfunction of the hydraulic system or the electrical system in one half of the unit, the corresponding system in the other half will operate alone with no loss of aircraft control. From a conventional, manual or electrical input, the dual systems operate as one to provide a single output. Separate modulating pistons, spool valves and power pistons are provided for each independent system. The two modulating pistons are connected by linkage to coaxial shafts which are held in fixed angular relationship by levers attached to a spring cartridge. Whenever hydraulic forces on the two modulating pistons vary to a sufficient degree, due to a malfunction of either the electrical system or the hydraulic system, to overcome the spring force in the cartridge, the spring will either shorten or extend and thus provide a motion to actuate a switch which indicates the asymmetry of the two systems.

A vernier-type adjustment is provided on one of the spool valves for the exact alignment of the inlet to the separate spool valve cylinders. The adjustment is accomplished by installing a movable outer sleeve on one valve. The sleeve itself includes a threaded portion having threads of a different lead on its inner and outer surface diameters. Rotation of the sleeve moves the inlets to the desired position over the spool valve. The vernier-type adjustment permits the accurate, rigid positioning of the inlet ports where distances of 0.001 inch are important for proper operation.

The features of the present invention which are believed to be novel are set forth with particularity in the appended claims. The present invention, both as to its organization and manner of operation, together with further objects and advantages thereof, may best be understood by reference to the following description, taken in connection with the accompanying drawings, in which:

FIGURE 1 is a perspective view showing somewhat schematically a dual-parallel servomechanism of the present invention with a sidewall removed to show internal construction;

FIGURE 2 is a partial plan view of the device of FIGURE 1 cutaway to illustrate interior portions thereof;

FIGURE 3 is a cross-sectional view taken along line 3—3 of FIGURE 2; and

FIGURE 4 is a longitudinal, cross-sectional view taken along line 4—4 of FIGURE 3 on an enlarged scale.

Referring again to the drawings and more particularly to FIGURE 1, the dual-parallel servomechanism constituting the present invention, generally designated **10**, is shown schematically as comprising a housing member **12** having parallel valve banks **14** and **16** which, for purposes of the schematic description, may be considered to be identical. Therefore, only the elements in bank **14** are shown in full.

A power piston **18** is slidably mounted in a piston chamber **20** near the bottom of bank **14** and is anchored at its ends to fixed members **22** so that relative movement between the housing member **12** and the power piston **18** is accomplished by having the housing member **12** slide upon the power piston **18**. A spool valve **24** having a slidable internal portion is mounted in a chamber **26** and is positionable to direct hydraulic fluid under pressure from chamber **26** through a first passageway **28** into chamber **20** on one side of piston **18** and through a second passageway **30** into chamber **20** on the opposite side of piston **18**.

Hydraulic fluid under pressure is supplied to the spool valve **24** through a fluid inlet port **32**, a passageway **34** and a fluid outlet port **35**.

A modulating piston **36** is slidably mounted in a chamber **38** and is positionable by hydraulic fluid which may flow to the chamber **38** on one side of piston **36** through a passageway **40** or to the chamber **38** on the other side of piston **36** through a passageway **42**, depending upon the position of an electrically actuated transfer valve **44** which controls the flow of hydraulic fluid through a pressurized fluid inlet **46**. The transfer valve **44** may be of any conventional type which automatically directs fluid to the modulating piston **36** when actuated by signals received through electrical leads **48** and **50**, which, in turn, are conventionally connected to a gyroscope, not shown.

The modulating piston **36** is linked to one end of a bell crank **52** and the spool valve **24** is linked to the other end of crank **52** which, in turn, is pivotally mounted on one end of a shaft **54**. The other end of shaft **54** is rigidly affixed to one end of a bell crank **56** which has its other end rigidly affixed to one end of a shaft **58**. The shaft **58** is rotatably mounted in journals **60** and **60*a*** which form rigid connections with the housing member **12**. An aircraft control lever or stick **62** is rigidly connected to shaft **58** so that crank **56** can be rotated by pulling lever **62**.

The spool valve **24** may be positioned manually with lever **62** or it may be positioned automatically by modulating piston **36**.

The stick **62** is also connected to corresponding elements in the bank **16** of housing member **12** through a crank **56*a*** which is rigidly affixed to the other end of shaft **58** and a shaft **54*a*** which connects crank **56*a*** to a crank **52*a***. The crank **52*a*** has one end linked to a spool valve **24*a*** and its other end linked to a modulating piston **36*a***. The spool valve **24*a*** and the modulating piston **36*a*** are slidably mounted in bank **16** in identical manner with the valve **24** and modulating piston **36** and are controlled by a transfer valve **44*a***.

Under normal operating conditions, both the banks **14** and **16** will receive identical signals from a common source. Assuming that a signal is received through electrical lead **50** which causes transfer valve **44** to direct fluid to chamber **38** through passageway **40** so that the fluid will exert a force on modulating piston **36** causing it to move to the left, as viewed in FIGURE 1. This will pivot bell crank **52** about shaft **54** in a counter-clockwise direction sliding spool valve **24** to the right. As spool valve **24** moves to the right, outlet port **35** is uncovered so that fluid under pressure will flow from passageway **34** through chamber **26** and passageway **28** into chamber **20** where the fluid exerts a force on the right hand side of power piston **18**, as viewed in FIGURE 1. Since power piston **18** is anchored, this force will cause housing member **12** to move to the right, as viewed in FIGURE 1. Since bell crank **52** is connected to housing member **12** through shaft **54**, bell crank **56** and shaft **58**, movement of housing **12** while stick **62** remains stationary will close outlet port **35** shutting off the flow of fluid to power cylinder **18**. The elements mounted in bank **16** will operate in an identical manner.

A pair of bell cranks **64** and **64*a*** each have one end rigidly affixed to shafts **66** and **66*a***, respectively, which are rotatably mounted in journals **68** and **68*a*** forming a rigid connection with housing member **12**. The other ends of bell cranks **64** and **64*a*** are linked to the modulating pistons **36** and **36*a*** through links **69** and **69*a***, respectively, so that linear motion of pistons **36** and **36*a*** results in an angular motion of shafts **66** and **66*a***. Bell cranks **70** and **70*a*** are rigidly affixed to the shafts **66** and **66*a***, respectively, and each has a pair of legs **72** and **74** mounted at right angles to each other. Each leg **72** is connected by links **75** to a transducer **76** which convert the angular motion of shafts **66** and **66*a*** into signals which are sent back to the transfer valves **44** and **44*a***, respectively, as feedback signals. The feedback signals tend to move the transfer valves in a direction that would tend to balance the respective modulating pistons to a central position. The arms **74** are linked together by means of a differential spring cartridge **78** having a microswitch **80**. The spring **82** of the differential spring cartridge **78** is designed in such a manner that, should either of the modulating pistons **36** or **36*a*** become depressurized, the spring **82** will cause the depressurized piston to follow exactly the movement of the pressurized piston. If, on the other hand, either system should malfunction so that the modulating pistons are not synchronized with one another, the bias of spring **82** is overcome tripping the microswitch **80** and cuts off the current to the solenoids **86** and **86*a***. A conventional spring (not shown) returns the solenoid and the respective transfer valve back to a de-energized position. A conventional signalling circuit (not shown) is coupled to the microswitch **80** and it concurrently signals the aircraft pilot to discontinue the automatic operation of servomechanism **10** by de-energizing solenoids in each bank, such as the solenoid **86** for bank **14**. This opens a by-pass valve in each bank, such as the by-pass valve **88** in bank **14**, so that the modulating pistons **36** and **36*a*** will be depressurized. The modulating pistons **36** and **36*a*** will then be returned to neutral positions by a centering spring cartridge **90** which has one end anchored to a fixed member **92** and its other end connected to one of the arms **74**. The spring **94** in cartridge **90** is firm enough to maintain the modulating pistons **36** and **36*a*** in a static condition so that the end of bell cranks **52** and **52*a*** which are linked to the pistons **36** and **36*a***, respectively, become fixed fulcrum points.

Under these conditions, if an input force is applied to stick **62**, moving it to the right, as viewed in FIGURE 1, bell cranks **56** and **56*a*** will rotate clockwise, swinging bell cranks **52** and **52*a*** in a clockwise direction. This causes spool valves **24** and **24*a*** to move to the right. This movement of spool valves **24** and **24*a*** uncovers their respective ports, such as the port **35** for valve **24**. Uncovering of port **35** permits fluid under pressure to flow through line **28** into chamber **20** on the right side of power cylinder **18** causing housing member **12** to move to the right, all as viewed in FIGURE 1, covering port **35**.

It is apparent that the system in bank **14** and the system in bank **16** and their linkage mechanism must be exactly synchronized for satisfactory operation of the servomechanism **10**. It is also apparent that conventional manufacturing tolerances will not permit the various systems to be exactly synchronized upon assembling the mechanism **10**. Therefore, as will appear hereinafter, one of the spool valves is provided with a vernier-type adjustment so that the system in one bank may be adjusted to operate identically with the system in the other bank.

Referring now to FIGURES 2–4, the parts which have been shown schematically in FIGURE 1 are shown in detail in bank **16** and will now be described by using the same reference numerals to refer to the parts generally.

The piston assembly **18*a*** is mounted in a chamber **20*a*** and includes a piston **100** and a piston rod **102** having its end connected to fixed members **22*a***. The piston **100** is provided with a piston ring **104**, a right hand piston face **106** and a left hand piston face **108**. A bearing **110** is mounted on rod **102** at each end of the chamber **20*a***, each bearing being maintained in position by means of retainers **111** having external threads **112** engaging internal threads **114** at the ends of chamber **20*a***. The bearings and retainers are conventionally sealed by O-rings and packing glands to prevent leakage of hydraulic fluid.

The spool valve **24*a***, most specifically illustrated in FIGURE 4, has conventional four-way porting and includes an encompassing sleeve **124** in which is provided a pressure port **125** communicating with port **35*a*** in bank **16**, a first power-cylinder port **126** communicating with passageway **30*a***, a second power-cylinder port **127** communicating with passageway **28*a***, and a fluid-return port **128** communicating with a sump **130** mounted in housing

12. A plurality of resilient seals **132** are employed to seal the various ports from one another and to seal the sleeve **124** in chamber **26a**. A cylindrical insert **134** is mounted in the sleeve **124** with one end abutting a retainer ring **136**. It is maintained in position by a threaded insert **137** having external threads **138** which engage internal threads **139** on sleeve **124**. A pressure port **140**, relief ports **142** and power-cylinder ports **144** are provided in insert **134** leading into a series of annuli adjacent the respective ports in sleeve **124**. Flow of hydraulic fluid through the power-cylinder ports **144** is controlled by a piston assembly **146** which is slidably mounted in insert **134**. The piston assembly **146** includes lands **148**, controlling flow of fluid through power-cylinder ports **144**, lapped balance grooves **150** and a passageway **152**. The passageway **152** extends through the piston **146** and communicates with the sump **130**. The piston assembly **146** also includes a collar **154** which is normally biased into engagement with a perforated plate **156** by a compression spring **158** having one end bearing against an end cap **160** and its other end bearing against a nut **162** which threadedly engages one end of piston assembly **146**. A fluid damper **164**, having an orifice **166**, is secured adjacent the perforated plate **156** by a perforated nut-plate **168** which threadedly engages the piston assembly **146** and is restrained against rotation by means of a friction plug **170**.

The cap **160** has an external thread **172** which has a first, predetermined lead and which threadedly engages the internal thread **174** in chamber **26a**. The cap **160** also has an internal thread **176** which has a second predetermined lead differing from the lead of the thread **172** and which threadedly engages an external thread **178** on sleeve **124**. A wrench-engaging surface **180** is provided on cap **160** so that a wrench may be employed to thread the valve **24a** into chamber **26a**. A dowel pin **182** may be inserted in an opening **184** in housing **12** to engage a slot **186** formed on sleeve **124**, thereby preventing rotation of the valve **24a** while permitting the sleeve **124** to slide longitudinally within chamber **26a** when cap **160** is subsequently rotated. Since the thread **172** has a first lead and the thread **176** has a second, different lead, the sleeve **124** and the cylindrical insert **134** carried by the sleeve **124** may be very finely adjusted relative to the lands **148** on piston assembly **146**. This is an important feature of the invention because it permits adjustment of the hydraulic system in bank **16** to exactly synchronize with the system in bank **14**. A friction plug **188** is mounted in sleeve **124** and engages the internal thread **176** on cap **160** restraining cap **160** against unwanted rotation by vibrational forces.

The modulating piston **36a** is mounted in a chamber **38a** and includes a piston **190** and a piston rod **192**. The piston **190** includes a ring groove **198** in which a piston ring **199** is mounted. The link **69b** connects one end **200** of piston rod **192** to the upper end of bell crank **52a**. A first conventionally sealed bushing **206** threadedly engages one end of chamber **38a**. A second similarly sealed bushing **214** is maintained in position at the other end of chamber **38a** by a retainer **216** which encompasses the end of the piston rod **192** and is abutted by an end cap **218**. The end cap **218** threadedly engages housing **12** and is sealed thereto with an O-ring.

Passageways **40a** and **42a** connect a pressure-fluid inlet port **46a** in valve **44a** with chamber **38a** adjacent opposite sides of piston **190**. A solenoid **86a**, a by-pass valve **88a**, and leads **48a** and **50a** are also provided.

The servomechanism **10** may be connected to an aircraft control, not shown, by means of a bearing **220** threadedly engaging housing **12** and which is prevented from rotating by a lock nut **224**.

Caps **226** and **226a** are employed to cover sumps **130** and **130a**, respectively and seals **228** and **228a** are mounted in the caps **226** and **226a**, respectively, to form a wiping connection with links **69a** and **69**, respectively.

While the particular dual-parallel servomechanism herein shown and described in detail is fully capable of attaining the objects and providing the advantages hereinbefore stated, it is to be understood that it is merely illustrative of the presently preferred embodiment of the invention and that no limitations are intended to the details of construction or design herein shown other than as defined in the appended claims.

What is claimed is:

1. A hydraulic servomechanism comprising:
a housing having a first and second power pistons and a first and second spool valves mounted therein,
a first and second passage means connecting the first and second spool valves to the first and second power pistons respectively, the spool valves controlling flow of hydraulic fluid to the power pistons,
a port means mounted in the housing for connecting the spool valves to a source of hydraulic fluid under pressure,
an input means mounted on the housing for positioning the spool valves simultaneously to direct hydraulic fluid under pressure to the power pistons through the passage means,
the input means including:
(a) first and second modulating pistons slidably mounted in said housing;
(b) third and fourth passage means mounted in said housing for connecting said first and second modulating pistons, respectively, to a source of hydraulic fluid under pressure;
(c) first and second valve means mounted in said housing for controlling the flow of fluid through said third and fourth passage means, respectively, in response to electrical input signals; and
(d) first and second linkage means connecting said first and second modulating pistons to said first and second spool valves, respectively.

2. The servomechanism of claim **1** wherein said input means comprises:
(a) a manual input lever pivotally mounted on said housing;
(b) first and second modulating pistons slidably mounted in said housing;
(c) third and fourth passage means mounted in said housing for placing said first and second modulating pistons, respectively, into communication with a source of hydraulic fluid under pressure;
(d) first and second valve means mounted in said housing for controlling flow of hydraulic fluid to said first and second modulating pistons, respectively, in response to input command signals; and
(e) first and second linkage means connecting said manual control lever to said first and second spool valves and to said first and second modulating pistons, respectively.

3. The servomechanism of claim **2** including third and fourth linkage means connecting said first and second modulating pistons, respectively, to feedback means for feeding a signal back to said first and second valve means.

4. The servomechanism of claim **3** including:
(a) differential spring cartridge means connecting said third and fourth linkage means together in such a manner that a predetermined force acting upon one of said modulating pistons is transmitted through said third and fourth linkage means and said differential spring cartridge to the other of said modulating pistons; and
(b) sensing means connected to said differential spring cartridge means for sensing a predetermined, unequal force transmitted by said modulating pistons through said third and fourth linkage means.

5. The servomechanism of claim **4** including a centering spring cartridge connected to said third and fourth linkage means for returning said modulating pistons to

a predetermined neutral position when they are not being acted upon by a hydraulic force.

6. The servomechanism of claim 5 wherein one of said spool valves contains a vernier-type adjustment means permitting said one spool valve to be synchronized for identical parallel performance with the other said spool valve.

7. In a dual-parallel servomechanism having modulating pistons adapted to operate in unison in response to command signals, means for detecting non-unison operation of said modulating pistons comprising:

(a) first linkage means connected to one of said modulating pistons and second linkage means connected to the other said modulating pistons;

(b) differential spring cartridge means connecting said first and second linkage means together, said differential spring cartridge means including a compression spring adapted to change its length upon non-unison operation of said modulating pistons; and

(c) sensing means connected to said spring for emitting a signal when said spring changes its length.

8. A hydraulic servomechanism comprising:

(a) a housing having first and second power pistons and first and second spool valves mounted therein;

(b) first and second passage means connecting said first and second spool valves to said first and second power pistons, respectively, said spool valves controlling flow of hydraulic fluid to said power pistons;

(c) a manual input lever mounted on said housing for positioning said spool valves simultaneously to direct hydraulic fluid under pressure to said power pistons through said passage means;

(d) first link means connecting said lever to said spool valves, first port means mounted in said housing for connecting said spool valves to a source of hydraulic fluid under pressure;

(e) means anchoring said power pistons whereby said housing slides on said power pistons when said hydraulic fluid is directed thereto through said passage means;

(f) first and second modulating pistons slidably mounted in said housing;

(g) third and fourth passage means mounted in said housing for connecting said first and second modulating pistons, respectively, to a source of hydraulic fluid under pressure;

(h) first and second electrical valve means mounted in said housing for controlling the flow of hydraulic fluid through said third and fourth passageways, respectively, in response to electrical input signals;

(i) third and fourth link means connecting said first and second modulating pistons to said first and second spool valves, respectively;

(j) fourth and fifth link means connecting said first and second modulating pistons, respectively, to electrical feedback means for feeding an electrical signal back to said first and second electrical valve means;

(k) differential spring cartridge means connecting said fourth and fifth link means together in such a manner that a predetermined force acting upon one of said modulating pistons will be transmitted through said fourth and fifth link means and said differential spring cartridge to the other of said modulating pistons;

(l) sensing means connected to said differential spring cartridge means for sensing a predetermined, unequal force transmitted by said modulating pistons through said fourth and fifth link means; and

(m) a centering spring cartridge connected to said fourth and fifth link means for returning said modulating pistons to a predetermined neutral position when they are not being acted upon by a hydraulic force.

9. The servomechanism of claim 8 wherein one of said spool valves contains a vernier-type adjustment means permitting said one spool valve to be synchronized for identical parallel performance with the other spool valve.

References Cited by the Examiner

UNITED STATES PATENTS

2,597,419	5/1952	Westbury et al.	91—216
2,773,660	12/1956	Rasmussen	91—216
2,826,896	3/1958	Glaze et al.	91—216
2,920,650	1/1960	Moog	137—625.69
2,942,583	6/1960	Rue	137—625.69
2,943,606	7/1960	Willis et al.	91—216
3,015,313	1/1962	Faisander	91—365
3,098,412	7/1963	Reitman	91—216

SAMUEL LEVINE, *Primary Examiner.*

FRED E. ENGELTHALER, *Examiner.*